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THE GEORGE W. WOODRUFF SCHOOL OF MECHANICAL ENGINEERING

Georgia Institute of Technology
Atlanta, Georgia 30332-0405

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HITCHING MECHANISM FOR LUNAR DUMP TRAIN

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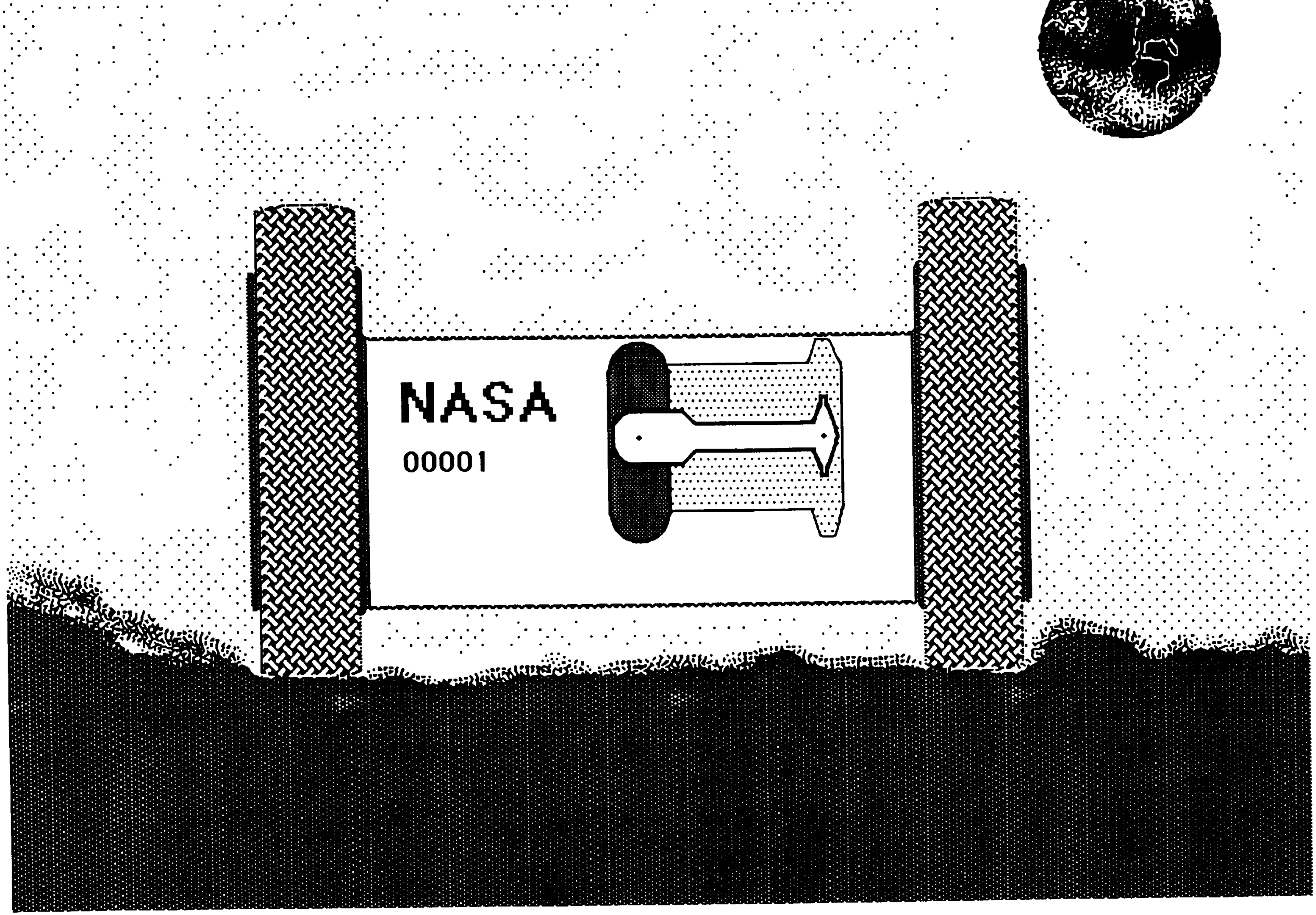
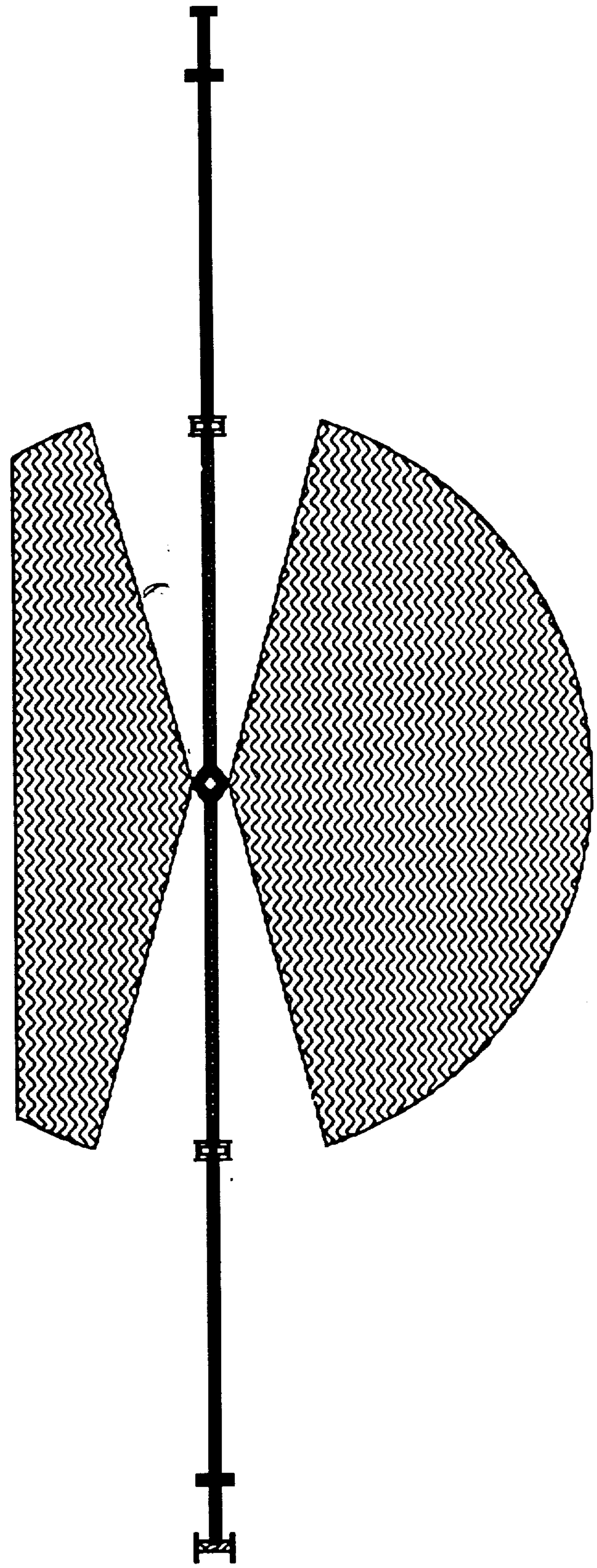
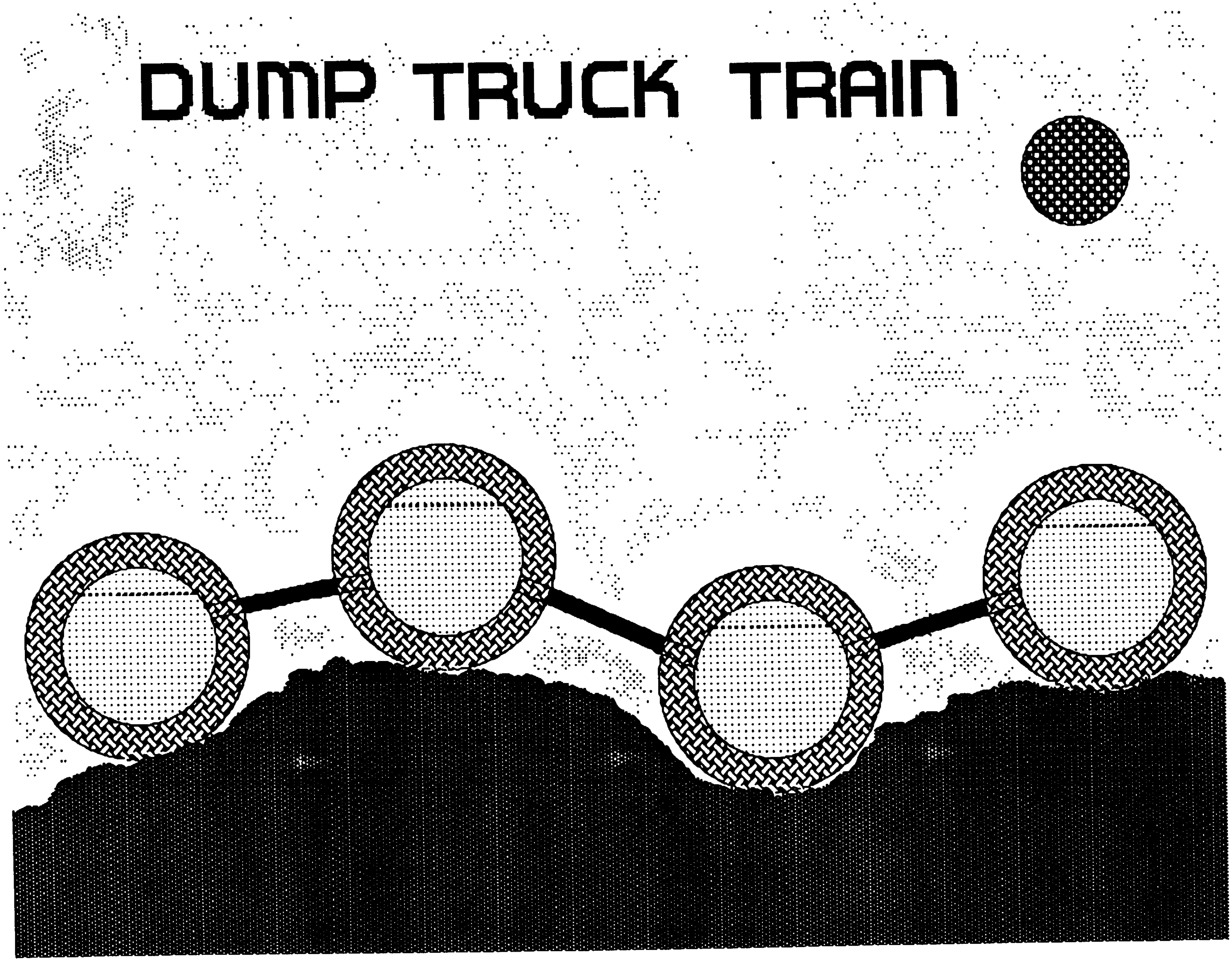
THOMAS G. HART
PETER M. SIMONSON
GREGORY Q. BROWN
ANDREW S. RING
TIMOTHY A. MARTIN
NILS C. NEWMAN

LUNAR DUMP TRUCK






DUMP TRUCK TRAIN

Abstract

The lunar dump hitch is a device to connect any number of the two-wheeled lunar dump trucks together into a train configuration. It consists of three major components: a flexible member, a boom, and a hitch mechanism. The flexible member is centered in a cutout of the dump bed and is designed to flex to provide the necessary articulation for the individual trucks to negotiate obstacles in the terrain. It is connected to a central pivot point and controlled by a stepping motor which serves the purpose of aligning the entire hitch mechanism for connection between two carts.

The boom folds out from a recession in the side of the cart to meet a boom from another cart and define a rigid common upright position for two connected dump beds. Some variation in this alignment is available due to the deflection of the flexible member. It is sized to withstand any forces up to and including collision between carts and collision of a train. It is connected to the flexible member by a hinge.

The hitch is a two piece unit with each cart having one male and one female end. It is designed to provide a rigid joint in the x-y plane while allowing motion in the x-z plane. It also provides a large target tolerance in the y direction.

Together these components form a hitch which allows articulated movement between individual carts in a train. The dump train hitch fully retracts so as not to interfere with normal operation of the trucks alone. It is easily deployed by a simple spin maneuver and can align itself for the connection procedure.

Table of Contents

1 Problem Statement

Background

A two wheeled dump module has been developed for use in lunar construction operations. The module is designed for remote operation. It is now desired to develop a hitching mechanism to link up to four modules together so that they may be operated as a single unit.

Objectives

The hitch is to have an automatic engaging/disengaging mechanism to eliminate the need for human labor. The mechanism must also allow passive operation of each individual module so that the entire train may be operated as a single unit. The train will be required to negotiate the uneven lunar terrain, including hills and tight turns. It is also necessary that the hitch may be stored so that it will not hinder operation of each module individually.

Constraints

The mechanism must adaptable to the lunar environment and have a long service life with no maintenance, yet it must be light enough to make transportation to the moon economically feasible.

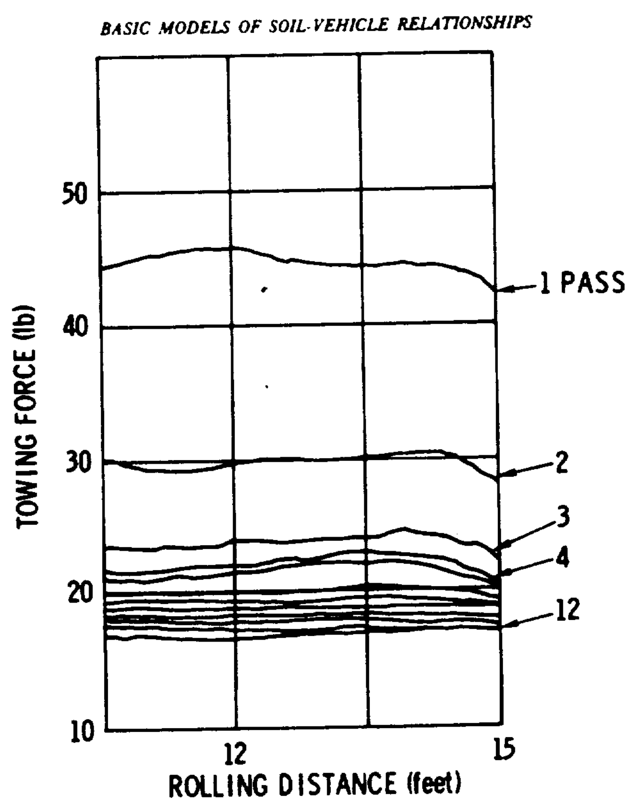
2 Movement as a Train

There are several benefits that can be obtained from connecting separate dump truck units into a single train. One of the more obvious benefits is the reduction in the number of separate controls required for the operation of the dump trucks. Another advantage is the reduction in soil compaction energy for dump truck movement. Of course, for either of these benefits to be realized, the separate units in the train must follow a common path.

The location of the hitch, which acts as the turning joint, was chosen to be midway between the axles of the dump trucks. This configuration was chosen because it produces the most stability at speeds of 10 mph or lower (Bekker p.545). This location of the joint also provides a good following capability for the train.

The motion of trailer trains has been studied extensively by Frederick Jindra of Southwest Research Institute. Jindra wrote an article which appears in volume 71 of SAE Transactions titled *Maneuverability of Trailer Trains.* In this article, he reduced by use of a four wheel steered tractor with delayed steering, the cut in and cut out of a 10 trailer train executing a 90 degree turn to less than 25 inches. Since each unit in the train being dealt with has steering capabilities, it is conceivable that each unit will be able to, with the proper controls system, follow in the path of the preceding unit almost exactly. This capability would give the maneuverability needed for navigation on the lunar surface and produce the invaluable effect of soil compaction.

Soil compaction, should prove to be a valuable feature produced by having the units of the train follow a common path. The effect on energy requirements due to compaction of dry sand which is similar on composition to the lunar surface is shown in Figure 1. As can be seen the



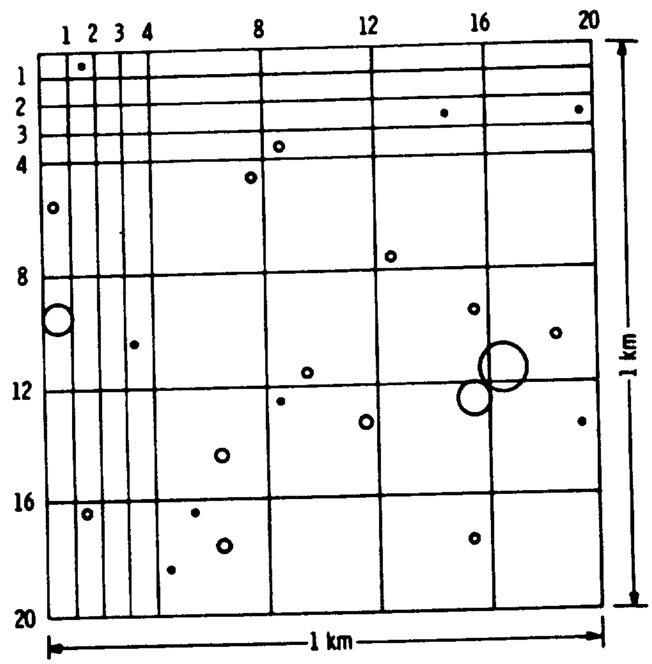
Towing force of a rigid wheel 30 in. diameter and 6.5 in. wide; load W = 300 lb; dry sand.

Figure 1

second dump module should encounter about 30% less rolling resistance than the first dump module and the rolling resistance of the fourth module is approximately half that of the first.

Analysis of pictures of the moon taken by Ranger VII has produced data on the density of craters on its surface. This density has been determined to a first approximation to be 23 craters per square kilometer. A random simulation of this density is shown in Figure 2.

PHYSICAL AND GEOMETRICAL TERRAIN VALUES



Distribution of the number and size of craters for $\rho = 23$ per km^2 simulated from a photograph by Ranger 7.

Figure 2

As can be seen these craters should be easily passed and should pose no difficulty in the train's movement. In fact, even if the density of the craters were ten times as great, i.e. 230 per km^2, the mean value distance between neighboring craters would be from 30 to 40 meters. Therefore, even in the denser areas the train, which will be approximately 3.2 meters wide, should still have no difficulty in lateral movement.

Even though craters should not be a factor in the train's movement, smaller obstacles and their effects on the motion of the train need to be considered. Simplification of natural obstacles is possible by classifying them as either of two elementary types: step down or step up. These two combined form either an embankment or a ditch. All natural shapes, as irregular as they may be, can be approximated with these two sets of two surface elements (Bekker, p.162).

There are only two basic types of failure to clear an obstacle (1) hang up failure (HUF), when the bottom of a vehicle, or in our case the connecting boom, interferes with the obstacle, and (2) nose in failure (NIF) when the front end of the vehicle interferes with the obstacle. This second failure mode, NIF, should not be a problem due to the fact that the dump trucks have no protruding parts in the front.

If the dump truck train experiences hang up failure, it will most likely occur when the train crosses a small steep embankment with a height greater than the clearance between the boom and the ground. An example of this type of HUF is shown in Figure 3 on the following page.

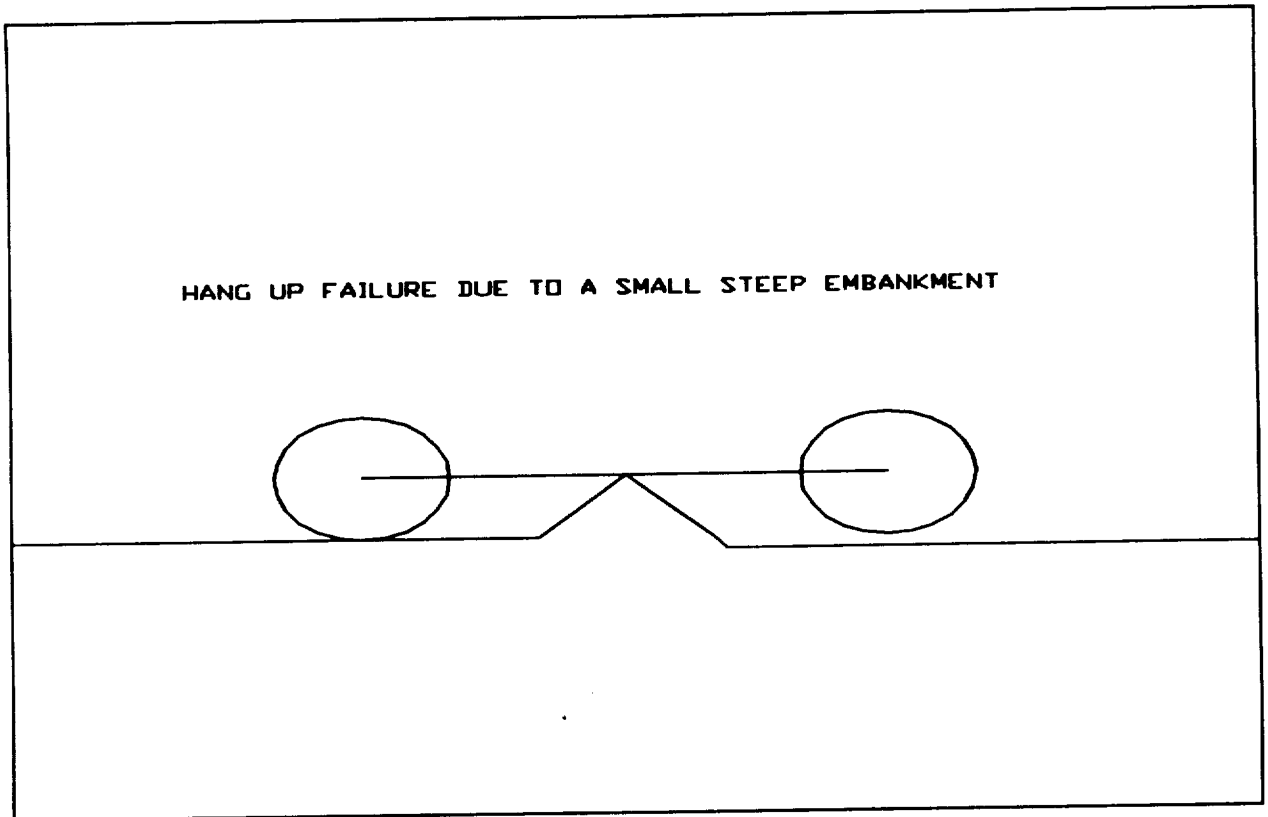


Figure 3

If this type or any other type of HUF occurs, it is a simple matter to disconnect the affected dump trucks and rejoin them to train after the obstacle has been bypassed.

3 Kinematics

3-1 Location of Ranges of Motion

The goal of the project is to find a balance between flexibility (range of motion), stability (enhancement of terrain negotiating capabilities), and simplicity (light weight and few moving parts). The first step in the determination of the required ranges of motion and placement of pivots is to define the types of motion. The ranges of motion are broken down into

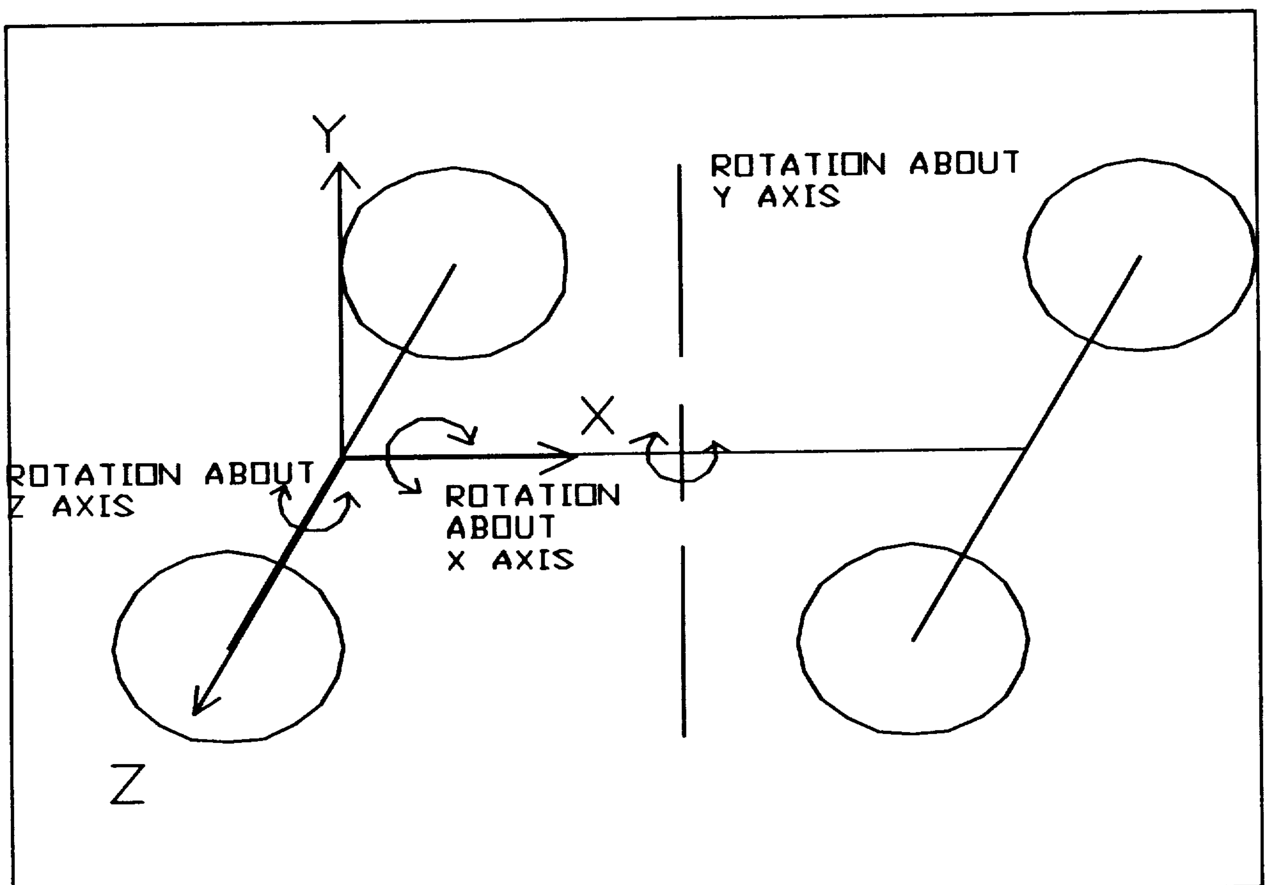


Figure 4

their six simplest forms: rotation about the X, Y, and Z axes, and translation along those axes.

Next, each axis must be defined to give an appropriate frame of reference. The Y axis is defined as the vertical axis (perpendicular to the ground). The Z axis is the axis of rotation of the wheels (parallel to the ground and along the length of the dump bucket). The x axis is the other axis parallel to the ground and along an imaginary line from the center of the dump bucket, perpendicular to the Y and Z axes. (refer to Figure 4)

A model analysis of the required ranges of motion displayed that all of the necessary motion can be obtained by the use of rotational elements. Because of the relative simplicity of rotational elements to translational elements and the required reliability and low maintenance, it is desirable to use only rotational elements.

It was determined that the optimal location for rotation about the Z axis is in-line with the axis of rotation of the wheels. This location will allow hill climbing capability which takes advantage of the natural self righting capability inherent in the bucket design. If this range of motion were located between the buckets instead of at the axle of the buckets, there would be the possibility of the pivot to allow the hitch to buckle and become wedged into the ground.

The optimal location for rotation about the X axis is on a line extending from the center of one bucket to the bucket that it is to be hitched to. This location allows any wheel to rise to roll over a rock or similar obstacle without causing the hitch to bind.

The best location for rotation about the Y axis is on the midpoint of a line extending from the center of one bucket to the center of the bucket to which it is to be hitched. This position allows for each bucket to follow the same path when negotiating a turn. There are two advantages to this type of travel. First, it allows negotiation of tight terrain without the tendency for each successive dump module to stray off into obstacles. Second, it is very advantageous due to the decreased rolling resistance of each successive bucket following the same path.

3-2 Sizing of Lengths

The criteria used to determine the length of the hitch boom is as follows; the vehicles when assembled as a train, must be able to turn relative to each other at a 45 degree angle. Thus, the boom must be of length great enough so that the tires will not rub when negotiating a turn of this angle. The hitch boom must also not be excessively long, or it may create ground clearance problems when negotiating uneven terrain. Taking these two factors into account, the boom was sized so that it would just allow the vehicles to turn relative to each other at a 45 degree angle.

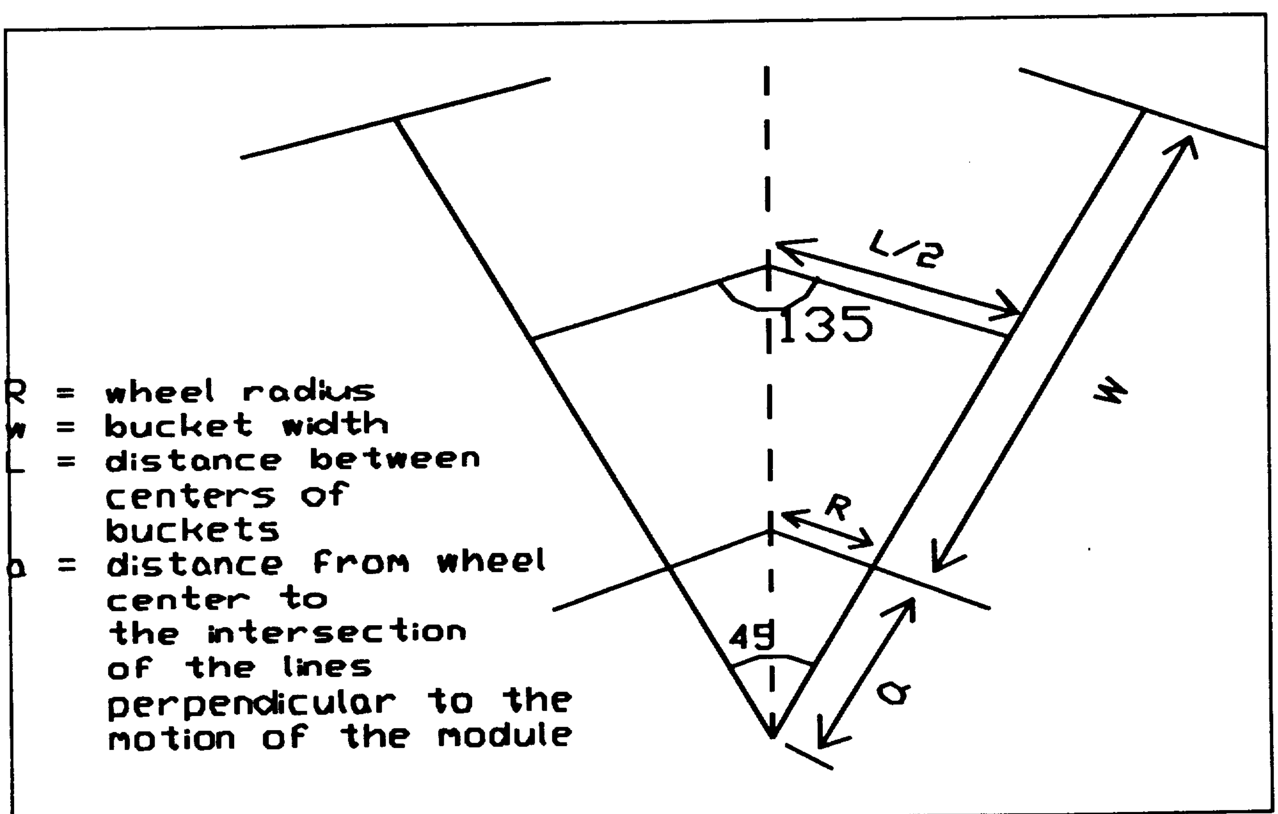


Figure 5

Given these constraints, the hitch length was determined in the following manner (refer to Figure 5 above);

$a = R/\tan(45/2) = 2.41R$

$L/2 = (a + w/2) * \tan(45/2)$

$L/2 = R + w/4.8$

The dump vehicle specifies that;

$R = 1.14\text{m}$

$w = 2.76\text{m}$

Therefore: $L/2 = 1.14 + (2.76/4.8)$

$L/2 = 1.7$

4 Component Description

4-1 Center Pivot

The entire hitch and boom is to be mounted on a center pivot which will allow rotation about the Z axis. The pivot is to be .2 meters wide and 8 centimeters in radius (see Appendix I for sizing). The pivot is to be housed in a cutout in the center of the bucket, and is to be constructed of the same AA 336 aluminum as the rest of the dump bucket. The pivot will only bear the load of the hitch, as the weight of the bucket itself will be supported at the center by the upper and lower braces of the cutout.

Housed in the center pivot will be a stepping motor that will allow for control of the rotational position of the hitch with respect to the bucket. A stepping motor was chosen because it can give accurate open loop position control, and is stable when stationary.

4-2 Deflection Beam - Flat Spring

Motion about the x-axis and torsion about the y-axis are achieved through the use of a 1.6 meter long flat spring (see Appendix II). The spring, which has an effective bending length of 0.52 meters on each side, is completely contained within the diameter of the bucket (see figure 7).

The deflection member is used to provide two directions of motion and self-righting capability with the least number of moving parts. The spring is supported at the center axis of the bucket through the center of gravity. The beam is affixed to the bucket by bearing mounts and a stepping motor. At the edge of the bucket, the beam is mounted to the collapsible exten-

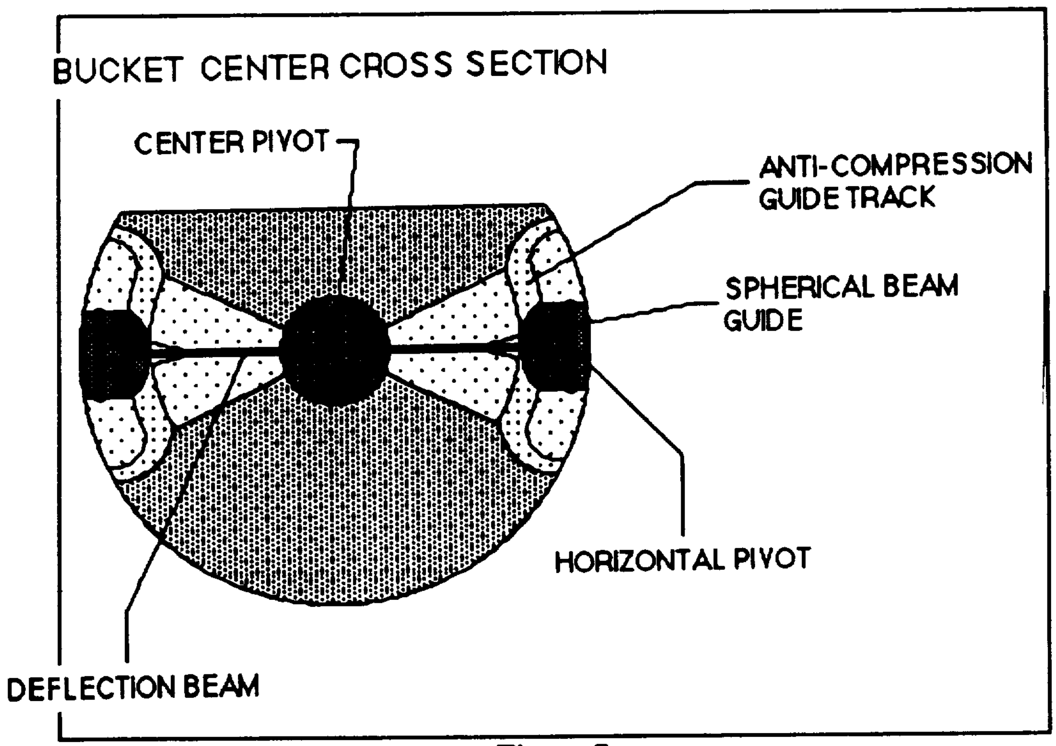


Figure 7

sion boom by a 90 degree horizontal pivot.

The beam is capable of a range of deflection of +/- 25 degrees in bending and +/- 45 degrees in torsion. The motion of deflection is guided by tracks at the horizontal pivot to prevent excessive axial compression of the deflection beam (see Figure 8). The beam is also able to pivot about the central axis of the truck. These criteria provide an acceptable range of motion for train performance.

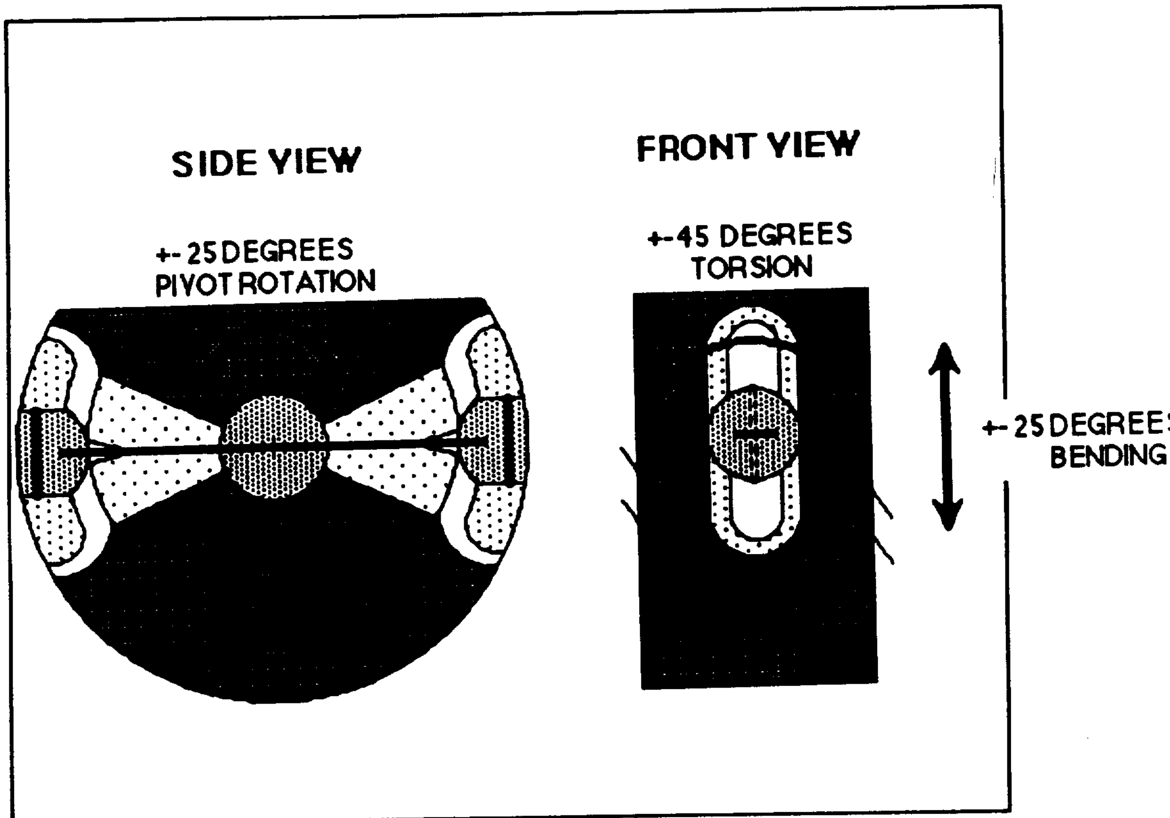


Figure 8

When unloaded, the deflection beam is stiff enough to semi-rigidly support 50 kilograms on the moon (more than the weight of the extension boom). This rigidity provides a means of alignment prior to hitching. The torsional capabilities also serve to align the hitch. The self-righting/self-aligning nature of the deflection beam is supplemented by the stepping motor to aid in final alignment prior to hitching, especially on uneven surfaces.

4-3 Hinge

The hinge connects the flexible member to the extendable boom. It must pass through a 90 degree range of motion to allow the boom to extend. The mechanism must be capable of locking in either the extended or retracted position, but when unlocked it must slide freely enough to allow the "spin" method of deployment of the boom. Additionally, it must fit into a channel in the bucket side to transfer any compressive forces on the boom to the sides of the bucket to prevent the flexible member from buckling. This channel will also serve to align the flexible member during torsional deflection.

The hinge consists of four pieces: female section, male section, pin, and slider. All but the slide are made of Al 6061 T6 aluminum. A brief description of each follows (See Appendix III for drawings).

This female section is pinned to the flexible member and joined to the male section by a pin. The slider mechanism with solenoid wiring is internal for the purpose of locking the male piece in either the 0 or 90 degree position. The female piece has a transition fit with the pin and is welded at both exterior surfaces.

This male section is welded to the circular boom on one side and pinned to the female section on the other. The center protrusion is slotted in two place to receive the slider locking mechanism. It has a clearance fit with the pin to allow it to pivot on the axis of the pin made of the same AL 6061 aluminum.

The pin is a circular rod that passes through the male and female sections and is fixed to the female section.

The slide is a flat rectangular metal of high shear strength and good electromagnetic properties. This indicates a low carbon, high silicon cold rolled steel. It is contained within the female section and is normally forced into the slots on the male section by two springs. When its surrounding coil is powered it is withdrawn totally into the female section.

The surfaces between the male and female sections and between the male section and the pin must be lubricated. A thin coating of a metal impregnated with MoS_2 is the best available dry lubricating system tested in literature (Space Materials Handbook, Chapter 9). These pieces will move only when the hitch is deploying and retracting so the overall number of cycles will be small.

The entire hinge must be heat shielded to prevent the temperature of the heat treated aluminum from exceeding 300 degrees fahrenheit and changing its properties. Also excessive temperature changes could cause thermal expansion problems at the interface of the female section and the flexible member.

4-4 Boom

The boom will be made of 6061 T6 aluminum. It will be a round member with a diameter of 14.4 cm. The boom will have a 1 cm. diameter hole through the middle to contain the wires needed for the hitch actuators. One end of the boom will be welded to the hinge which connects it to the fexible member inside the dumptruck. The hitch will be welded to the other end of the boom. The boom will fold into a recession in the side of the dump truck. See Appendix VIII for stress analysis.

4-5 Hitch

The hitch has a male and a female part. The two parts act as a hinge joint for rotation around the y axis of the vehicle. The hitch also acts as a connecting mechanism for the dump trucks. Both parts of the hitch are made from aluminum 6262 (T9) for wear, lubricant, weld, and strength properties. The male part is welded to the end of one boom and the female part is welded to the end of the other boom on each dump truck. The male part is shaped like two circular cones with small cylinders on each end. The female part has a hole and a latch for connection and rotation. The female part rotates around each small cylinder of the male part. For illustrations of the hitch refer to Appendix VI. For a more detailed analysis of the hitch refer to Appendix VII.

5 Component Functions

5-1 Extension and collapse

The hitching mechanism is to be deployed from the retracted position by utilizing the capability of the dump module to rotate about it's X-axis. This motion will allow the boom to be extended due to inertia.

First the locking pin at point A (see Figure 9) is released, freeing

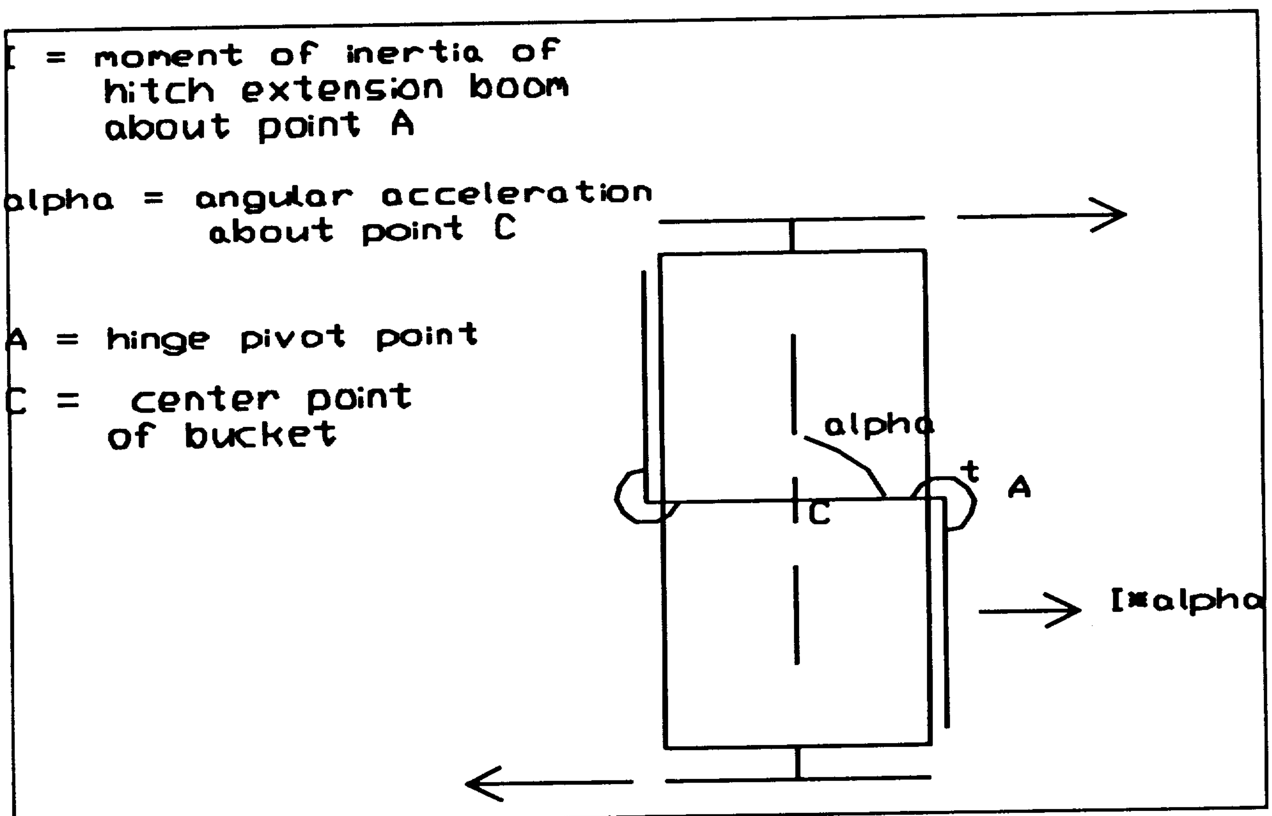


Figure 9

the boom to rotate about point A. Each wheel is then quickly accelerated in opposite directions to produce an angular acceleration about the center of the bucket (point C on Figure 9). The wheels are suddenly halted, causing an inertial force of the boom, I * Alpha, which overcomes the torque, t, caused by the friction of the hinge at point A, causing the boom to rotate outward and deploy. Once in the fully extended position, the locking pin at point A is inserted into a channel in the hinge to lock the boom in place.

This design for deployment has a backup system inherent in the capabilities of the dumptruck itself. If a hinge should become obstructed or jammed, the boom may be deployed by locking the hitch to an extended hitch on another module, and pulling the boom outward to the extended position.

5-2 Alignment

In order to mate the male hitch of one vehicle to the female hitch of the next vehicle, the position of both members to be connected must be controlled in three planes. First it is necessary that both hitch members are aligned on the vertical axis, that is to say, thay are at the same height. This positioning is achieved using the synchronized logistical capability of the stepping motor located at the center pivot of the hitch. Because the hitch is inherently balanced at the center pivot, it may be rotated about this axis by incrementing the stepping motor in the pivot point, using the weight of the bucket as a counterbalancer. This rotation yields a vertical displacement at the end of the boom where the hitching member lies. Once vertical alignment is achieved, alignment on the plane parallel to the ground may be achieved by utilizing the existing mobility characteristics of the dump module.

5-3 Hitch - Locking and Unlocking

The hitch is aligned using the stepping motors and the wheels of the dump trucks. The latch on the female part is opened using the actuator after the male part is released. The male part of the hitch is aligned with the hole in the female part of the hitch. When the male hitch part falls in the hole the latches close. Finally, the actuator is closed and the latching is complete.

Separating the dump trucks is accomplished by releasing the actuator, and raising the male hitch using the stepping motor. This causes the latch to open releasing the male and the female parts of the hitch. For illustrations of the hitch refer to Appendix VII.

6 Shielding

For protection from radiation, micrometeoroids, and abrasion, the following areas need some form of shielding:

Electrical:

- hitch actuator
- servo control
- sensors at hitch point
- sensors at servo

Mechanical:

- servo bearings
- deflection beam sweep
- horizontal pivot
- hitch actuator

Micrometeoroid shielding of both mechanical and electrical systems can be achieved through multilayer foils or wire meshes (example-nickel/inconel mesh). These forms of shielding are particularly necessary for the deflection beam to prevent pitting which would greatly excel fatigue cracking. The electrical systems are shielded from micrometeoroids and radiation. Wires routed to the truck from the hitch, run through the center of the boom to provide maximum protection. The mechanical systems at the friction points are protected by teflon, brush systems, and soft metals. Abrasion will be a major problem since the impact of micrometeoriods tends to positively charge surfaces thus attracting dust.

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Appendix I - Center Piviot Specifications

Sizing of the center pivot Given the width of the flexible center member as 0.13 meters, the width of the center channel is to be .2 meters, to allow for clearance and dust accumulation on the flexible member. The strength criteria used is that the weight of all four dump modules must be able to be suspended from the center pivot of one module, to give an approximate simulation of a head on collision. Thus, the load encountered by the pivot is; dump weight * 4 = 12,700N*4 = 51kN

Because the moment at the center of the module is supported by the upper and lower braces of the cutout, the loading of the center pivot is similar to that of a center loaded beam, fixed at both ends. The equation for the moment at the center of such a loaded member is (ref. Shigley, p. 810);

$M = F*L/8$

Given $L = 0.2m$, $F = 51kN$

Thus; $M = 1.3kN*m$

Given the fatigue strength for aluminum; $S = 34.5$ MPa

A safety factor of 10 gives a maximum allowable stress of; 3.45 MPa

Given the formula to find the stress of a member loaded in bending

Sigma = M*c/I

Where I for a member of circular cross section is $(pi*r^4)/4$

And the maximum stress is found at c = r

Thus; Sigma = $4*M/(pi*r^3)$

And r = $(4*M/(pi*Sigma))^{1/3}$

This gives a value of r = 0.08m. It is recommended that the surface be hardened or coated with hardened steel to resist abrasion.

Appendix II - Deflection Beam Specifications

Criteria:

- Must be able to deflect 25 degrees at worst case over 0.52 meters.
- Must be able to deflect 45 degrees in torsion over 0.52 meters.
- Must be able to support up to 50 kg semi-ridgidly and deflect fully under a load of 6700 kg.
- Must be able to survive 6700 kg inertal forces.
- Thermal expansion must not interfere with hinges and pivots.
- As light weight as possible.
- Must have good fatigue resistance.
- Must have good abrasion resistance or good coatablity.

Choices:

- Beryllium Copper
- Composite
- Spring Steels AISI 9254-5

Calculations

Beam is 1.6 meters long.

The central pivot has a radius of 0.089 meters. Pivot is cut away to 0.04 radius to allow some deflection. The extension boom horizontal pivot and the mounting region occupy 0.24 meters.

$$0.8\ \text{m} - 0.04\ \text{m} - 0.24\ \text{m} = 0.52\ \text{m}$$

The effective spring length on each side is 0.52 meters. The stress due to bending at the end of the deflection beam

Unloaded

(lunar weight) 1.14 m/2 * 50 kg * 9.8/6 m/s = 46 N

(6 * 46 N * 0.52 m)/(0.13 m * 0.008 m 2) = $1.72 \cdot 10^7$ Pa

unloaded stress = $1.72 \cdot 10^7$ Pa

Maximum allowable stress for cross section = $4 \cdot 10^8$ Pa if AISI 9255 is used(footnot). This is a rough approximation which takes into account number of cycles, temperature, and other factors. It is presented only as justification of the design.

Loaded

(lunar weight) 1.14 m * 6745 kg * 9.8/6 m/s = 12560 N

(6 * 12560 N * 0.52 m)/(0.13 m * 0.008 m^2) = $4.7 \cdot 10^9$ Pa

loaded stress = $4.7 \cdot 10^9$ Pa

Thus the beam will fully deflect under the load of the cart producing good terrain following characteristics and the need for a curved support to protect the spring at maximum deflection.

When the stop is reached the stress profile is assumed to pure tension.

Maximum tension

$12560\ N/(0.13\ m * 0.008\ m) = 1.2*10^7\ Pa$

Which does not exceed the maximum allowable stress of $4*10^8$ Pa.

Torsional Stress:

Unloaded

When unloaded the torsional forces on the deflection boom are extremely low since the only source of torsion is the hitch. The low level of torsion allows for the self-righting capabilities.

Loaded

An approximate value of the applied torque from the next cart is

$5000\ N * 1.6\ m \sim= 8000\ N$

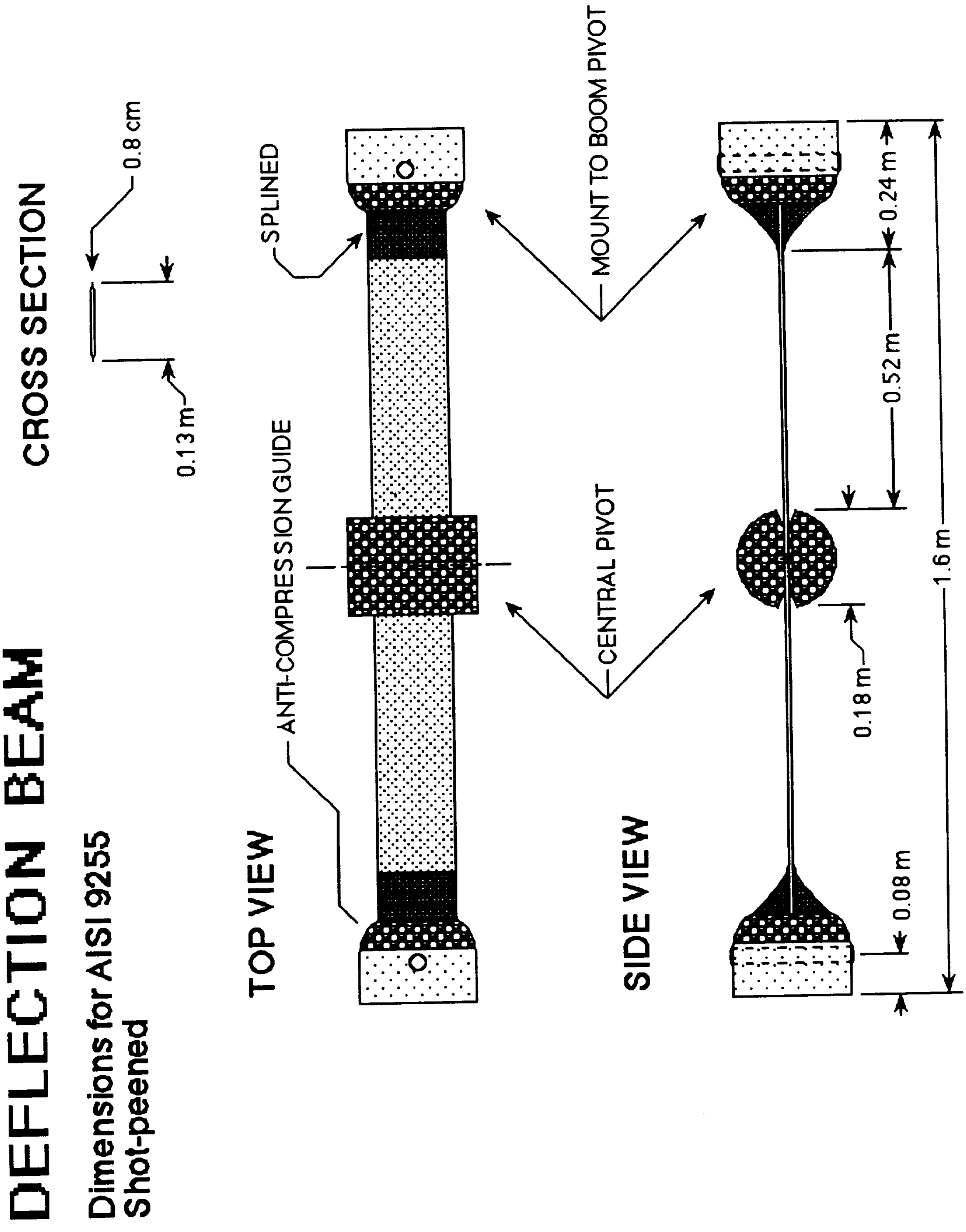
From St. Venant's approximation

$K2 = 1/(3*(1.0 + 0.6 * 0.008\ m * 0.13\ m)) = 0.32$

$(8000\ N * 0.52\ m)/(0.32 * 0.13\ m * 0.008\ m^2) = 1.5*10^9\ Pa$

This stress is greater than the maximum allowable stress, however this condition will not be experienced unless the cart exceeds the stated 45 degree angle of tilt. Thus the system produces rapid response providing good terrain following capabilities. The cutout in the cart, as in bending, would have to be cut to provide maximum support during maximum deflection.

This calculation is also only a general guideline. Flat steel springs are usually not recommened for torsional purposes. The cross-sectional area used in the calculations was rectangular. Other cross-sections such as elliptical would probably more appropiate. Furthermore, the choice of material varies greatly. Composites would be more appropiate strength wise and much lighter (with AISI 9255 the beam would weigh 13 kg), however little reliable information is available on long term behavior of composites in a lunar environment.

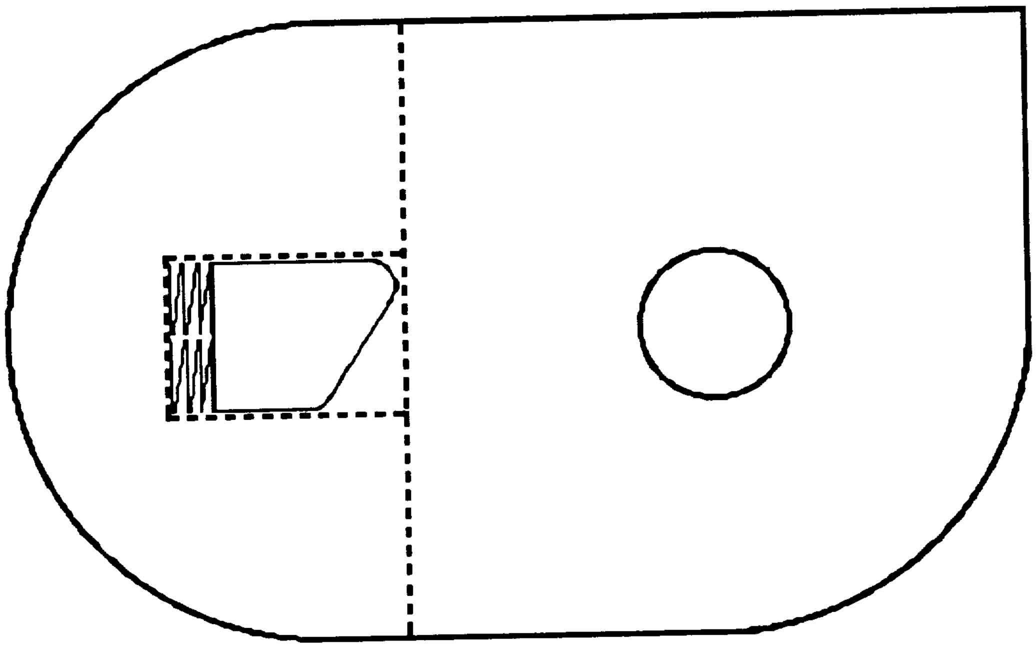

DEFLECTION BEAM
Dimensions for AISI 9255
Shot-peened
CROSS SECTION
0.8 cm
0.13 m
TOP VIEW
SPLINED
ANTI-COMPRESSION GUIDE
MOUNT TO BOOM PIVOT
CENTRAL PIVOT
SIDE VIEW
0.24 m
0.52 m
0.18 m
1.6 m
0.08 m

Appendix 3: Figure A

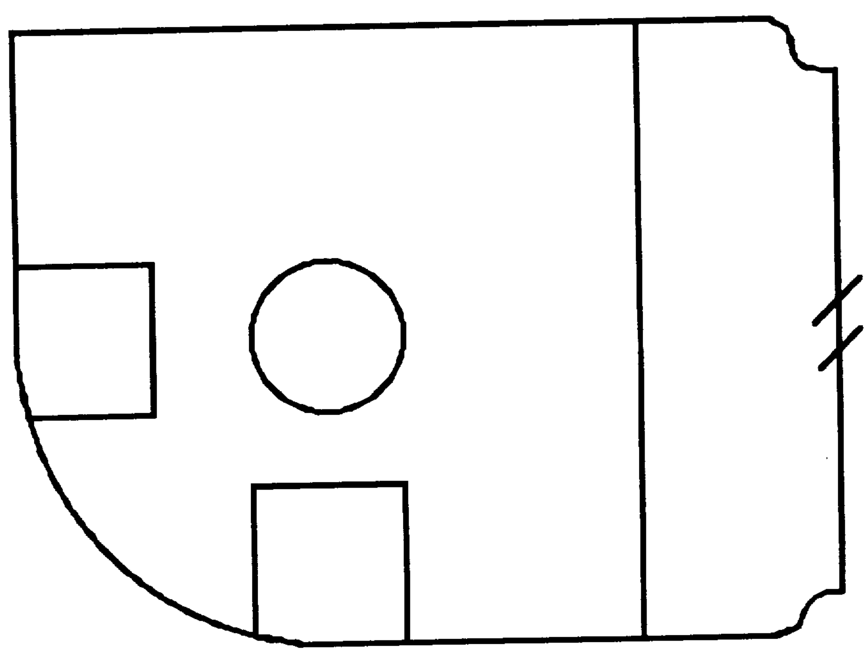
Scale = 50% actual size

HINGE: TOP VIEW

FEMALE SECTION



MALE SECTION

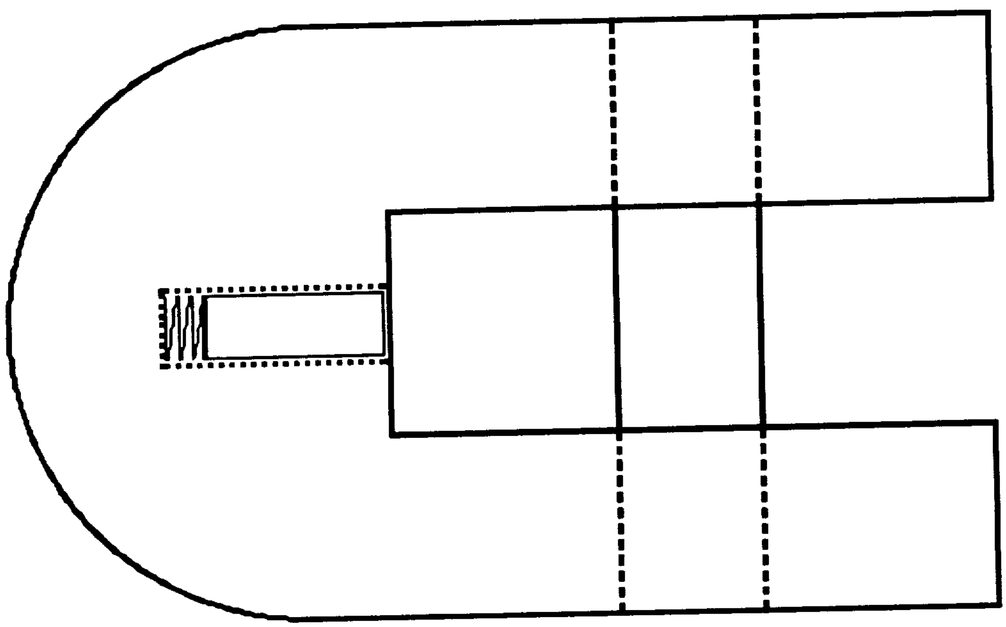


Appendix 3: Figure B

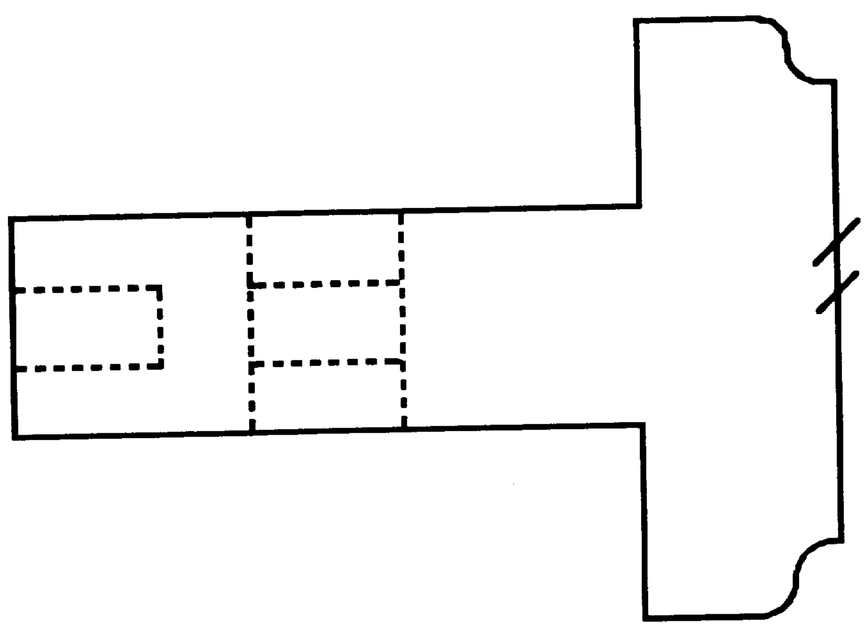
Scale = 50% actual size

HINGE: SIDE VIEW

FEMALE SECTION



MALE SECTION



Appendix IV - General Stress Analysis

Modes of material failure of the stiff member and hitch assembly.

- Maximum bending moment configuration
- Jerking of the vehicle
- Maximum force caused by impact
- Maximum force in tension

The maximum bending moment configuration occurs when one of the dump trucks is inadvertently driven off a cliff. This configuration puts the joiner in its greatest state of stress by having it hold the weight equal to one dump truck. The stress at each point is different and the stress calculations are included in each section of this report.

The maximum force caused by jerking of the vehicle occurs when one dump truck is standing still and the attached truck is at maximum acceleration.

$F = m*a$

$m = 6695$ kg

$a = .762$ m/s^2

$F = 5000$ N

The joiner is able to withstand the impact of colliding one dump truck with an immovable object. For this analysis it is assumed that the maximum number of dump trucks in the train configuration is four. The time of impact is assumed to be .05 seconds. The maximum force on a joiner from impact is equal to:

$F*dt = m*V$

$dt = .05$ sec (maximum time of impact)

$m = 3*6695$ kg (maximum vehicle mass on the joiner)

$V = 15$ km/hr (maximum velocity of a dump truck)

$V = 4.2$ m/s

$F(.05) = 3(6695)(4.2)$

$F = 2x10^6$ N

The final mode of material failure occurs when one vehicle is held in tension over a cliff. The maximum force on the joiner in this configuration is equal to the weight of one loaded dump truck.

$m = 6695$ kg

$g = 1.63$ N/kg

$F = m*g$

$F = 6695(1.63)$

$F = 10000$ N

In summary: 1) the maximum axial force on the joiner (which occurs on impact) is $2x10^6$ N; however, 2) the maximum bending moment is calculated separately for each individual critical point on the joiner.

Appendix V - Boom Stress Analysis

Determining the radius of the stiff members: The stiff member is designed around the modes of material failure of the stiff member and hitch assembly. The member cross section is circular, which allows for uniform properties when the member is rotated. Aluminum 6061 T6 is used because of its good machinability, weldability, and the presence of silicon which allows for easy forming. The maximum yield strength for Al 6061 T6 is 40 kpsi. The fatigue strength is 14 kpsi. The shear strength is 30 kpsi (Materials Selector, 1986). Material failure mode of maximum bending moment configuration for a circular cross section:

Maximum stress = Mc/I

$I = pi*r^4/4$

Maximum stress = $4Mr/pi*r^4$

Maximum moment occurred at l = 2.6 m

Maximum weight of dump truck W = 10000 N

Maximum moment = Wl = 30000 N*m

Maximum stress = $14x10^3$ psi*$6.89x10^3$ Pa/psi

Maximum stress = 96 MPa

$96x10^6 = 4(30000)/pi*r^3$

r = .07m

r = 7 cm

Material failure mode of maximum bending moment configuration for a circular cross section in shear stress:

Maximum shear stress = 4V/3A

V = 10000 N

$A = pi*r^2$

Maximum shear stress = $(30x10^3$ psi$)(6.89x10^3$ Pa/psi$)$ = 206 MPa

$206x10^6 = 4(10000)/3*pi*r^2$

$r^2 = 2.1x10^{-5}$

r = .0045 m

Material failure mode of maximum force caused by impact:

Maximum axial stress = F/A

$F = 2x10^6$ N

Maximum axial stress = $45x10^3$ psi$(6.89x10^3$ Pa/psi$)$

Maximum axial stress = $3x10^8$ Pa

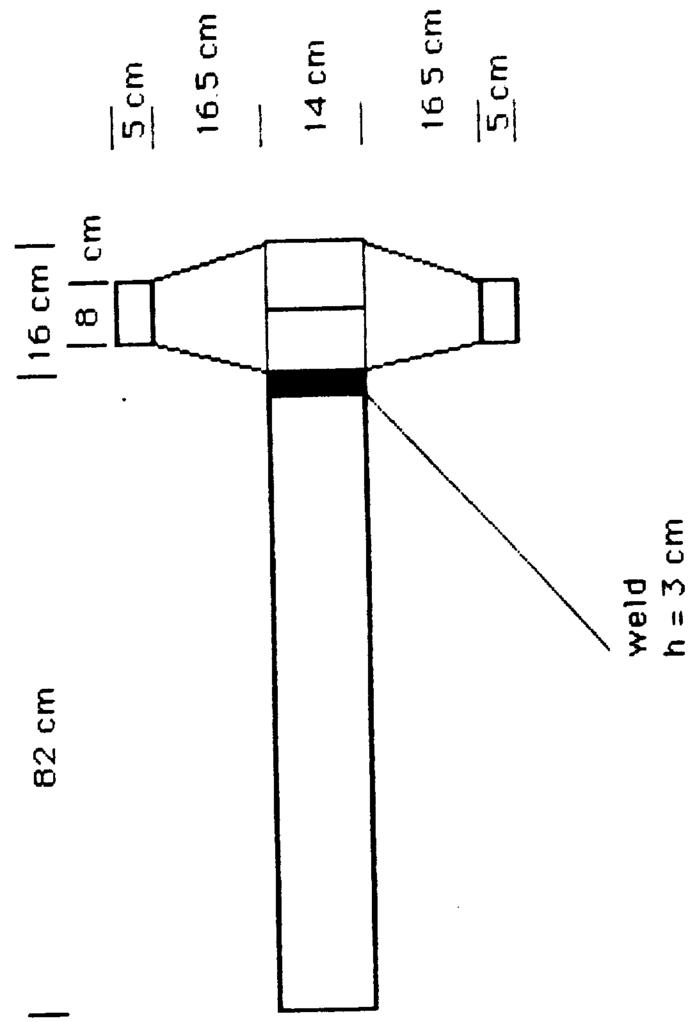
$3x10^8 = 2x10^6/pi*r^2$

r = .05m

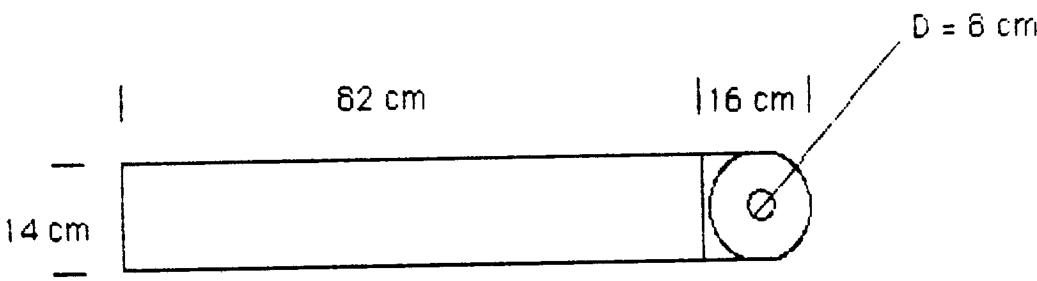
In summary the radius of the stiff member, necessary to keep the joiner from failing is 7 cm. The radius will stay constant from one dump truck to the next.

Appendix 6: Figure A

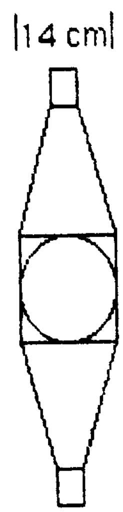
MALE HITCH AND BOOM
SIDE VIEW



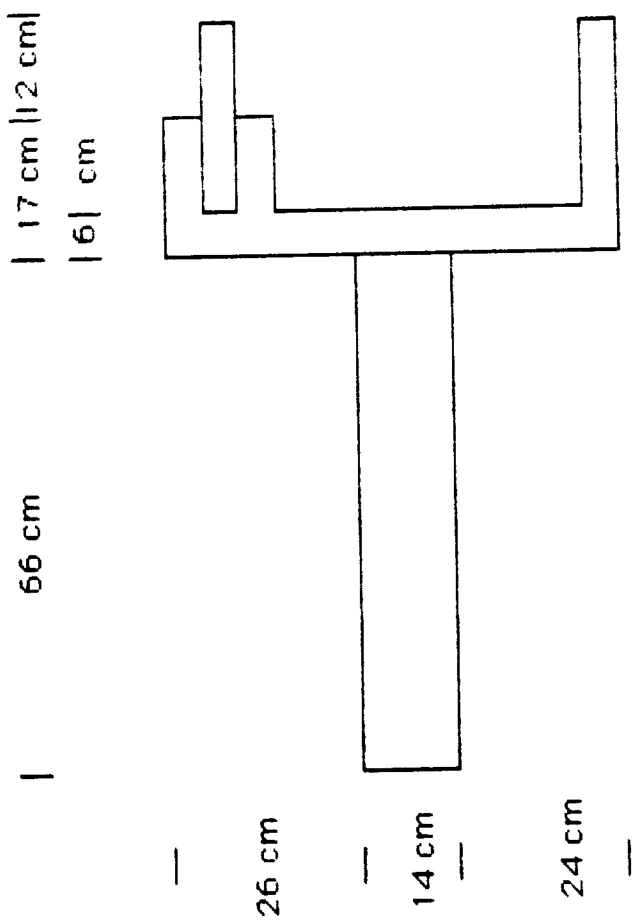
MALE HITCH AND BOOM
TOP VIEW



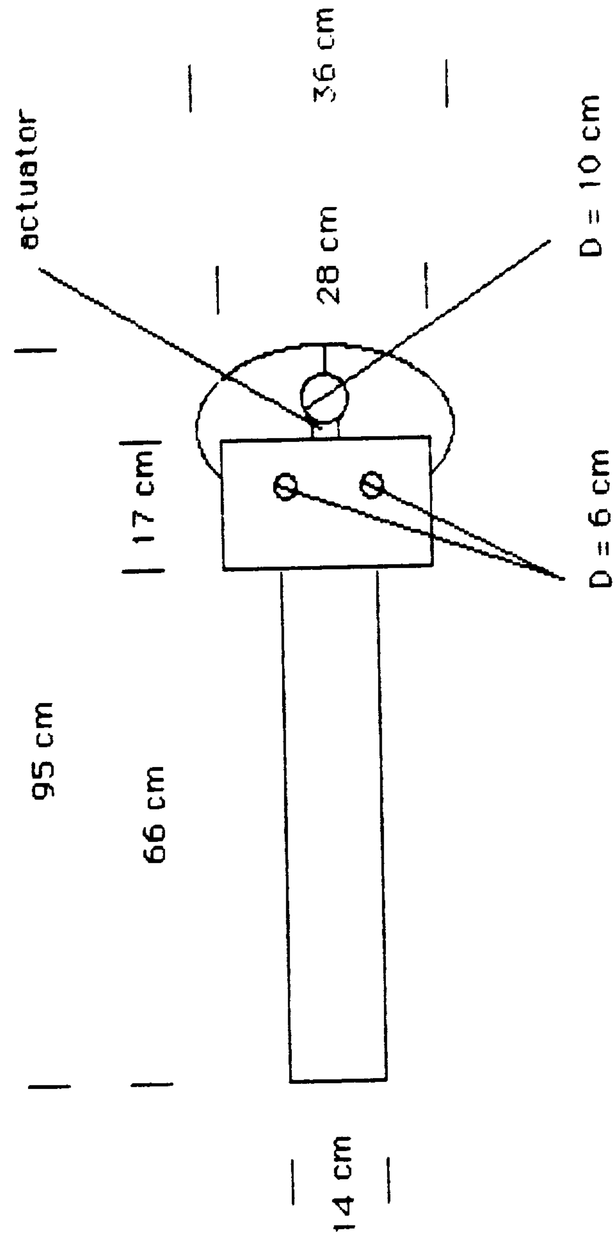
MALE HITCH
REAR VIEW



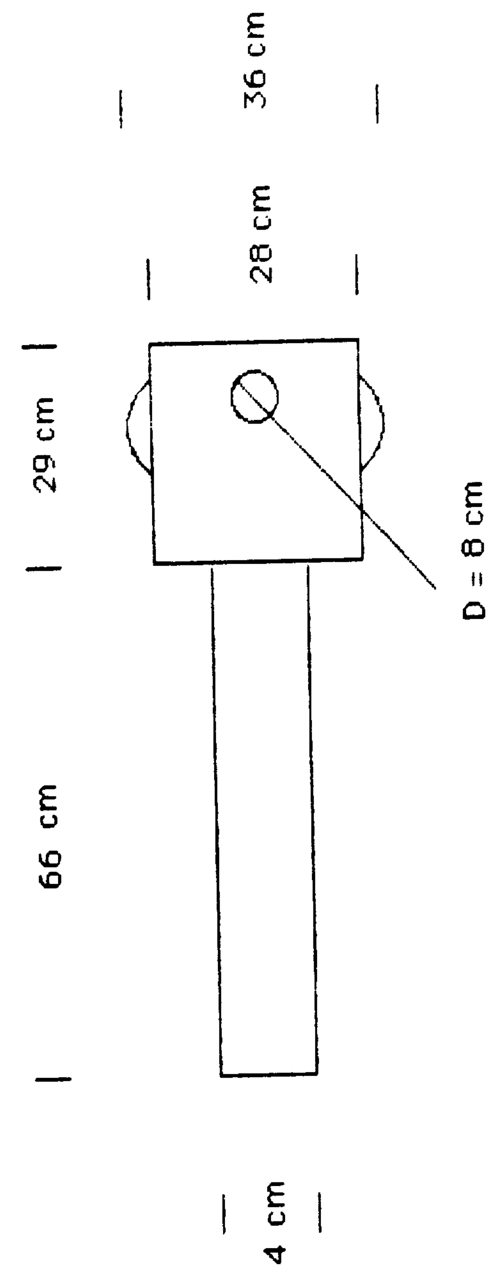
FEMALE HITCH AND BOOM
SIDE VIEW



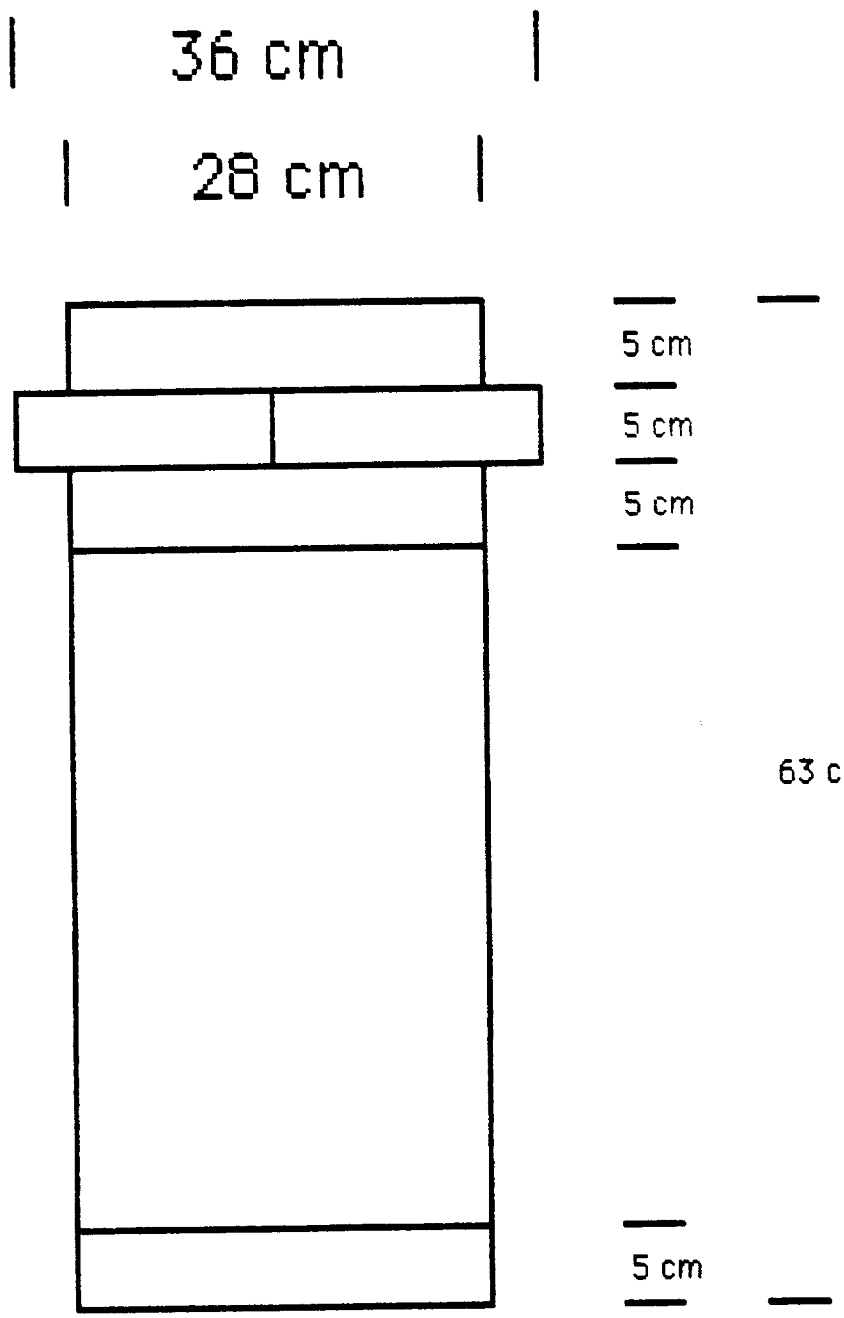
FEMALE HITCH AND BOOM
TOP



FEMALE HITCH AND BOOM
BOTTOM



FEMALE HITCH
FRONT



Appendix VII - Hitch Stress Analysis

Determining the radius of the male hitch: The male hitch is made from AL 6262 T9. The yield strength is 55 kpsi, the hardness is 210 Brinell, the endurance limit is 13 kpsi, and the shear strength is 35 kpsi. The hitch is welded to the end of one of the dump truck's booms. This material is used because it contains lead that acts as a lubricant for the hitch parts. The alloy also contains silicon which allows for easy forming. The material has high hardness, strength, and weldability. The maximum radius to keep from failing is determined from the impact force analysis. Maximum axial force equals 2×10^6 N. This force is divided in half because the top and bottom areas of the hitch are assumed to receive the same impact.

$$V = 1X10^6\ N$$

$$\text{Maximum shear stress} = 35\ X10^3\ \text{psi}\ (6.89X10^3\ \text{Pa/psi})$$

$$\text{Maximum shear stress} = 240\ \text{MPa}$$

$$\text{Maximum shear stress} = 4V/3A$$

$$\text{Area} = pi * r^2$$

$$240\ X10^6 = 4\ (1X10^6) / 3\ (pi * r^2)$$

$$r = .04\ m$$

$$r = 4\ cm$$

Determining the measurements of the female hitch: For the same reason outlined in the male hitch, the female hitch is formed out of AL 6262 T9 and welded to the other boom on each dump truck.

Determining the radius of the female hitch pins: The maximum shear force occurs on the pins during impact. Again, the top and the bottom areas of the hitch are assumed to receive the same impact. Thus, cross sectional areas of the male hitch should equal the total cross sectional area of both pins.

$$\text{pi}\, r_h^2 = 2\text{pi}\, r_p^2$$

$$r_h^2 = 2r_p^2$$

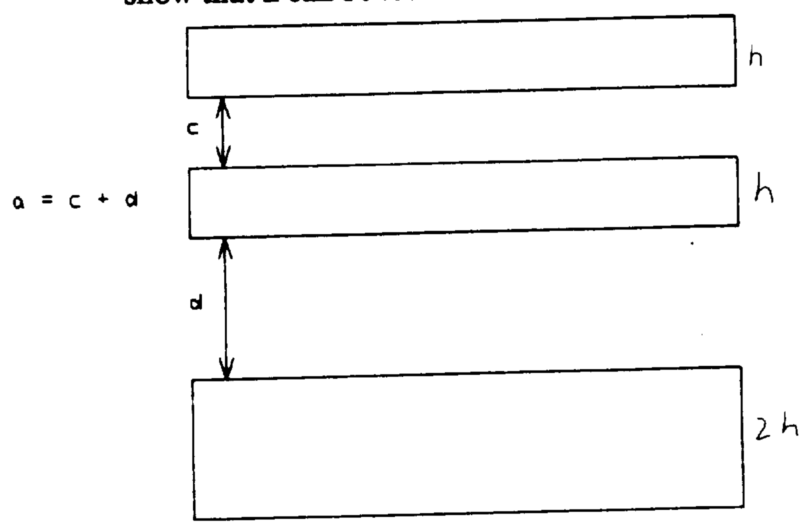
$$r_h = 4 \text{ cm}$$

$$r_p^2 = (.04)^2/2$$

$$r_p = .03 \text{ m}$$

$$r_p = 3 \text{ cm}$$

Determining the cross sectional area of the female hitch: The height of the female hitch is 57 cm. The base of the hitch needs to be 28 cm to give adequate hitching ability. The height of the hitching area needs to be at least 1 cm to keep from grinding into the male hitch. An approximation to the stress on this type of configuration follows and shows that it is appreciably less than the fatigue strength. Since the base is 28 cm and the pins are 6 cm in diameter and the hitch is 8 cm in diameter, the minimum material base is 8 cm. For the following configuration, a is determined to show that h can be less than 1 cm.



Maximum stress = Mc/I

M = 1.7 m * 10,000 N = 17,000 N m

Maximum stress = 13×10^3 psi * 6.89×10^3 Pa/psi

Maximum stress = 90 MPa

$I = bh^3/12$

$I = .08(.57^{)3}/12 - .08(a)^3/12$

I = .08/12 (.1852-a3)

$90 \times 10^6 = 17 \times 10^3 * .29/.08/12(.1852-a^3)$

a = .56 m

$h = .25$ cm

Thus, the cross sectional is left up to the designer.

Determining the heights of the welds: Material failure mode of maximum force caused by impact:

Maximum shear stress = F/.707*h*l

l = 2pi*r

r = radius of stiff member

r = 7 cm

Maximum shear stress of the material = 206 MPa

$F = 2 \times 10^6$ N

$206 \times 10^6 = 2 \times 10^6/.707*h*2*pi*.07$

h = 3 cm

Material failure mode from maximum bending moment configuration:

Maximum stress = Mc/I

$Iu = pi*r^3$

I = .707 * h * Iu

$I = .707 *h*pi*r^3$

M = 1.7 * 10,000 = 17,000 Nm

Maximum stress = 96×10^6 Pa

c = r = .07 m

$$96 \text{ X } 10^6 = 17{,}000/.707*h*pi(.07)^2$$

$h = 2$ cm

Thus ,h equals 3 cm for welding the male and female hitches to the booms.

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SCHOOL OF MECHANICAL ENGINEERING

ME 4182 MECHANICAL DESIGN ENGINEERING
WINTER QUARTER 1989

HITCHING MECHANISM FOR LUNAR DUMP TRAIN

WEEKLY PROGRESS REPORT

TEAM #8
THOMAS HART
RUSTY MARTIN
NILS NEWMAN
ANDREW RING
PETE SIMONSON
GREG BROWN

MEETING DATE: JANUARY 11,1989

WEEKLY GOAL: Determine a suitable project title

ACCOMPLISHMENTS: The group met on Tuesday to discuss the project title and discuss initial design considerations. The project title was agreed to be "Hitching Mechanism for Lunar Dump Train".

Some of the problems considered were;

- The engaging/disengaging mechanism,
- Pivot points: relationship to centers of mass,
- How to store the hitches when not in use,
  * Hitches as independent units
  * The hitch as an integral part of the dump module
- Power distribution within the train,
- Hill and turn negotiation,
- Ground clearance during dumping.

Nils contacted the T.A. to request bluprints for the dump module so that we may begin experimenting with possible kinematic design configurations.

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WEEKLY PROGRESS REPORT
TEAM #8

THOMAS HART, RUSTY MARTIN, NILS NEWMAN
ANDREW RING, PETE SIMONSON, GREG BROWN

MEETING DATE: JANUARY 18,1989

WEEKLY GOAL: Develop rough draft for problem statement.

ACCOMPLISHMENTS: The team met on Monday and discussed the hitching and pivot mechanisms, and the problems and conditions associated with operation in the lunar atmosphere.

On Tuesday we met in the design lab in the French Building. There we dicussed design and prototypes with Mr. Brazell, and developed a graphic displaying the vehicle tracking path using Versacad.

Each team member searched the on-line databases available in the library for appropriate reference sources.

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WEEKLY PROGRESS REPORT

TEAM #8

THOMAS HART, RUSTY MARTIN, NILS NEWMAN
ANDREW RING, PETE SIMONSON, GREG BROWN

MEETING DATE: JANUARY 25, 1989

WEEKLY GOAL: Submit problem statement and break up group to handle separate tasks.

ACCOMPLISHMENTS: This week the group was broken up into subgroups and each subgroup or individual was given an assignment. Nils will be investigating which materials will be most appropriate for the lunar application. As of now, everyone else will be investigating the kinematics of the hitching mechanism and the problem of deployment/idle-storage of the hitch. Each subgroup will submit ideas to the main group for feedback and refinement of ideas. Once a general kinematic setup has been decided upon, Greg and Pete will refine the kinematics and Rusty and Andrew will solve the dynamic equations of the vehicles. Tom will investigate the applications of CAD software for the project and continue to handle the administrative tasks of the group.

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HITCHING MECHANISM FOR LUNAR DUMP TRAIN

WEEKLY PROGRESS REPORT
TEAM #8
THOMAS HART, RUSTY MARTIN, NILS NEWMAN
ANDREW RING, PETE SIMONSON, GREG BROWN

MEETING DATE: FEBRUARY 1, 1989

WEEKLY GOAL: Define degrees of freedom and locate optimal placement and types of joints.

ACCOMPLISHMENTS: We broke down each possible type of joint and discussed it's avantages/disadvantages, and all of the degrees of freedom allowable by any specific joint. We then discussed the degrees of freedom as they affect the movement of the dump train, and discussed the optimal placement of joints.

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HITCHING MECHANISM FOR LUNAR DUMP TRAIN

WEEKLY PROGRESS REPORT

TEAM #9
THOMAS HART, RUSTY MARTIN, NILS NEUMAN
ANDREW RING, PETE SIMONSON, GREG BROWN

MEETING DATE: FEBRUARY 8, 1989

WEEKLY GOAL: Decide on the best method of boom deployment, and decide on the length of the boom. The optimal method of deployment should take into consideration that all of the hitch is prone to soil submersion, and it should retract sufficiently so as not to hamper single unit operation. We should now be deciding on material selection for the boom.

ACCOMPLISHMENTS: This week we formulated the decision matrix and decided to go with the center-mounted extending boom to minimize complexity. We came up with the concept of the counterweighted retracting boom. This idea has one complete hitch, and one hitch receptacle on each individual dump module. The hitch is counterweighted so that when it retracts it sticks straight up in the air, and when it extends it raises the center of mass so that the boom will fall onto the receiving end of the next module. We are now working on the best possible deployment/retraction method.

Also this week, we reviewed our design process up to present to prepare for our midterm report.

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HITCHING MECHANISM FOR LUNAR DUMP TRAIN

WEEKLY PROGRESS REPORT
TEAM #8

THOMAS HART, RUSTY MARTIN, NILS NEWMAN
ANDREW RING, PETE SIMONSON, GREG BROWN

MEETING DATE: FEBRUARY 15, 1989

WEEKLY GOAL: Select materials, design pivot points and hitch point, and develop a method of deployment that minimizes the need for additional actuators.

ACCOMPLISHMENTS: We came up with the idea of a center mounted hitch where the boom is retracted into the side of the dump module. This configuration has fewer moving parts than the telescoping mechanism, and it is able to use the existing motors on the modules to deploy the hitch. Another advantage of this configuration is that the pin which the hitch pivots on is not a load bearing member on the dump bucket, which gives us increased reliability. Also, the bucket is used as a counterweight, eliminating the need for the extra weight of a boom mounted counterweight.

We came up with the equation that the optimum length of the hitch (the length from axle to axle) is equal to the total width of the cart plus the diameter of the wheel.

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WEEKLY PROGRESS REPORT

TEAM #8

THOMAS HART, RUSTY MARTIN, NILS NEUMAN
ANDREW RING, PETE SIMONSON, GREG BROWN

MEETING DATE: FEBRUARY 22, 1989

WEEKLY GOAL: Develop rough draft for report, each individual should finalize his assigned design section.

ACCOMPLISHMENTS: Nils is investigating materials for use in the flexible center member, and has talked with Dr. Newman about flexible steels and copper alloys. Greg has researched patents for ideas on the hitching mechanism, and he and Peter are designing the hitching mechanism and center pivot points. Rusty has researched shielding for the pivot points and members that are susceptible to solar radiation. Andrew is working with Rusty on a design for the deployment mechanism and central pivot. Tom is investigating types of actuators and position sensors for aiding the deployment/retraction of the hitching mechanism, and starting to develop an outline for a rough draft.